Proceedings of the Fourth (1994) International Offshore and Polar Engineering Conference
Osaka, Japan, April 10-15, 1994

ISBN 1-880653-10-9 (Set); ISBN 1-880653-13-3 (Vol. III)

Oscillating Foils for Marine Propulsion

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ABSTRACT

The use of oscillating fins by aquatic animals for propulsion has inspired the design of corresponding propulsors for ships. Rigid and partly flexible oscillating propulsor designs were analyzed for a large ship based on two-dimensional linear and nonlinear theories. Open water efficiencies of the oscillating propulsors were 17-25% higher than those of an optimal screw propeller. Behind the ship, quasi-propulsive efficiency of the partly chordwise flexible foil was 72%, i.e. 5% higher than that of the screw propeller. The efficiency increase produced by the oscillating propulsor was attributed to the considerable increase in the propeller working area due to the wide span of the foil.

KEY WORDS: bio-propulsion, design, flexible foil, hydrodynamic design, propulsor, oscillating foil

NOMENCLATURE

C = Foil Chord Length
C_T = Thrust Coefficient = (Thrust)/{($\rho V_a^2/2$)·(C·S)}
C_{po} = Power Coefficient = (Power)/{($\rho V_a^3/2$)·(C·S)}
E = Young's Modulus
E^* = Nondimensional Young's Modulus = $E/(\rho V_a^2/2)$
h = Heaving Amplitude
J = Foil Advance Coefficient = $V_a/(n \cdot C)$
N = Foil Frequency (c.p.m. = cycle per minute)
n = Foil Frequency (c.p.s. = cycle per second)
S = Foil Span Width
t = Thrust Deduction Fraction
V_a = Foil Advance Speed
V_s = Ship Speed
w = Wake Fraction
α = Pitching Amplitude
ε = Feathering Parameter = $(\alpha \cdot V_a)/(h \cdot 2\pi n)$
η = Quasi-Propulsive Efficiency = (EHP)/(DHP) = $\eta_o \times \eta_r \times \eta_h$
η_o = Open Water Efficiency = C_T/C_{po}
η_h = Hull Efficiency = $(1-t)/(1-w)$
η_r = Relative Rotative Efficiency
ρ = Water Density
θ = Phase Delay Angle of Pitching to Heaving

INTRODUCTION

The method of propulsion evolved by many swimming animals is centered around the generation of thrust from movements of a crescent shaped fin or wing-type surface (van Dam, 1987). This includes propulsion from the tail fins of certain fish and marine mammals, carangiform (Lighthill, 1970) and thunniform (Hoar and Randall, 1978) propulsion, and also the flapping propulsion exhibited by the pectoral flippers of many marine turtle and penguin species. Maximum propulsive efficiency for a fin whale (*Balaenoptera physalus*) has been estimated to be about 85% (Bose and Lien, 1989). For a naval architect, natural propulsive fin geometries (Lang, 1966; van Oossanen and van Oosterveld, 1989; Bose et al., 1990; Curren, 1992) and their propulsive characteristics are important as guides in design of oscillating propulsors.

On the other hand, several theories of oscillating propulsion have been developed (Bose, 1992; Chopra, 1976; Kambe, 1978; Katz and Weihs, 1978; Kubota et al., 1984; Kudo et al., 1984; Lighthill, 1960, 1970; Liu and Bose, 1993; Wu, 1961,1971). Some of them have been extended to consider propulsion using wave energy (Bose and Lien, 1990; Lai et al., 1993; Isshiki and Murakami, 1982-1984; Isshiki and Murakami, 1986; Wu, 1972). Of these studies, Kudo et al. (1984) and Kubota et al. (1984) developed two-dimensional linear and nonlinear theories which can estimate propulsive performance of partly flexible as well as rigid oscillating propulsors. These show that a chordwise flexible foil has a higher propulsive efficiency than a similar rigid foil; that although linear theory overpredicts mean thrust and efficiency, it correctly predicts trends and thereby can produce a design chart to find an optimal motion of the foil; and that predictions of propulsive performance from nonlinear theory agree closely with experimental performance of oscillating foils if corrections are made for three-dimensionality of real foils and viscous drag. Hence, nonlinear theory can be used to predict the performance of a designed oscillating propulsor accurately.

This paper briefly describes the shape of cetacean species' flukes and their performance. Then rigid and partly flexible oscillating propulsor designs are analyzed for a large ship, a 200,000 DWT tanker, and compared with an optimal screw propeller design.

PROPULSIVE PERFORMANCE OF WHALE FLUKES

Bose et al. (1990) measured the bodies and flukes of 9 cetacean species; Figure 1 shows the planforms. The fin whale's flukes (fw) had the largest aspect ratio of 5.8 and the second lowest sweep angle of 30.7 deg.; the flukes of the white-sided dolphin (wsd) had the smallest aspect ratio of 2.7 and the highest sweep angle of 47.4 deg.; and the white whale's flukes (ww) had the moderate aspect ratio of 3.3 and the lowest sweep angle of 28.3 deg. Lui and Bose (1993) applied a quasi-vortex-lattice method based on small amplitude unsteady lifting surface theory for a rigid three-dimensional foil to

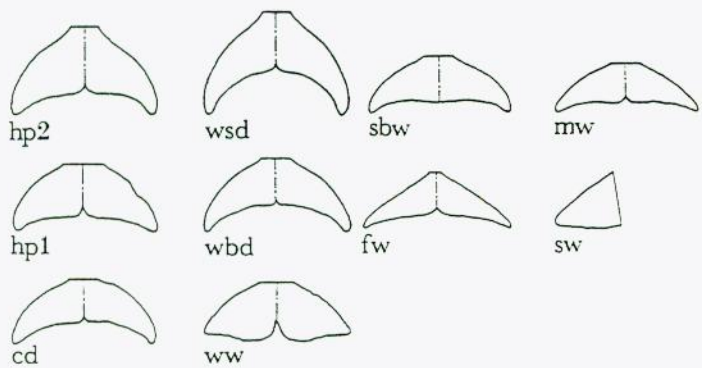

Fig. 1 Approximate shapes of fluke planforms (The key to the abbreviations is as follows: hp2, harbour porpoise 2; wsd, white-sided dolphin; hp1, harbour porpoise 1; wbd, white-beaked dolphin; cd, common dolphin; ww, white whale; sbw, Sowerby's beaked whale; mw, minke whale; fw, fin whale; sw, sperm whale.) from Bose et al. (1990)

Table 1 Estimated efficiencies of the fin whale's flukes

Speed (knots)	7.8	11.7	15.6	19.5	23.3
Efficiency	0.855	0.870	0.870	0.845	0.840

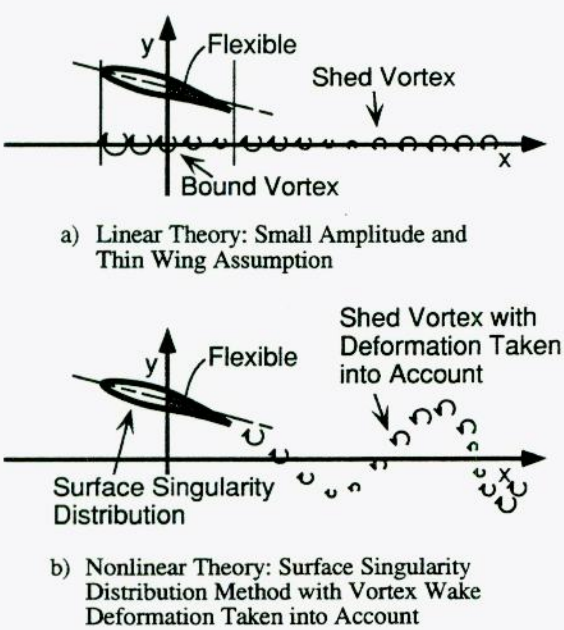

Fig. 2 Two-dimensional oscillating foil theories used for the design

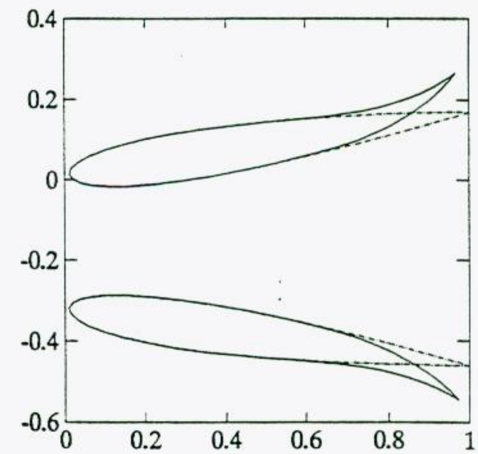

Fig. 3 The rigid (chain dotted) and flexible section shapes on the downstroke (upper) and upstroke for a foil oscillating at a reduced frequency of 0.2 and a feathering parameter of 0.4. The rear half of the foil is flexible; amplitude of the trailing edge deflection is 0.1 of the chord length.

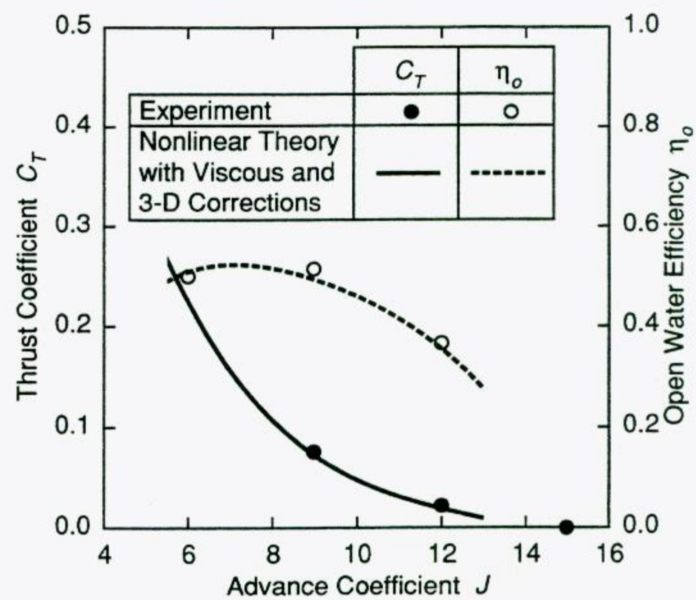

Fig. 4 Comparison of thrust coefficient and open water efficiency between experiment by Scherer (1968) and nonlinear theory with viscous and three-dimensional corrections (Kubota et al., 1984); Rectangular planform, Aspect ratio = 3, Section shape = NACA63A015, h/C = 0.6, θ = 90 deg., α = 10 deg., Pitching axis = 25% chord station

analyze the propulsive performance of these three propulsors. The results clarified several characteristics of three dimensional oscillating foil propulsors, but viscous effects, wing flexibility and large amplitude effects should be taken into account to predict the propulsive performance more accurately.

Bose and Lien (1989) estimated the propulsive performance of the fin whale's flukes using a two-dimensional small amplitude rigid foil theory with corrections for three-dimensionality, viscous drag and large amplitude effects. Table 1 shows the estimated efficiency (open water efficiency) of the flukes. The span of the flukes was 3.0 m; the heave amplitude was 1.5 m; and the tail-beat frequency was 3 rad/s. Pitch angle was chosen for the best efficiency. Very high efficiencies of more than 85% were predicted.

DESIGN OF OSCILLATING PROPULSORS FOR A 200,000 TON TANKER

Kudo et al. (1984) and Kubota et al. (1984) developed two-dimensional linear and nonlinear theories for a partly flexible foil as well as a rigid foil. The linear theory was a small amplitude thin wing theory, and the nonlinear theory was a surface singularity distribution method with the deformation of the vortex wake taken into account (Figure 2). Their results can be summarized as follows:

(1) It was shown theoretically that a partly flexible foil was more efficient than a rigid one. Figure 3 (Bose, 1992) demonstrates the geometrical effects of flexibility. The flexible part (rear half) deflects so that the hydrodynamic force is reduced. If the foil motion is the same, therefore, a flexible foil produces less thrust than a rigid foil. However, the deformation rotates the

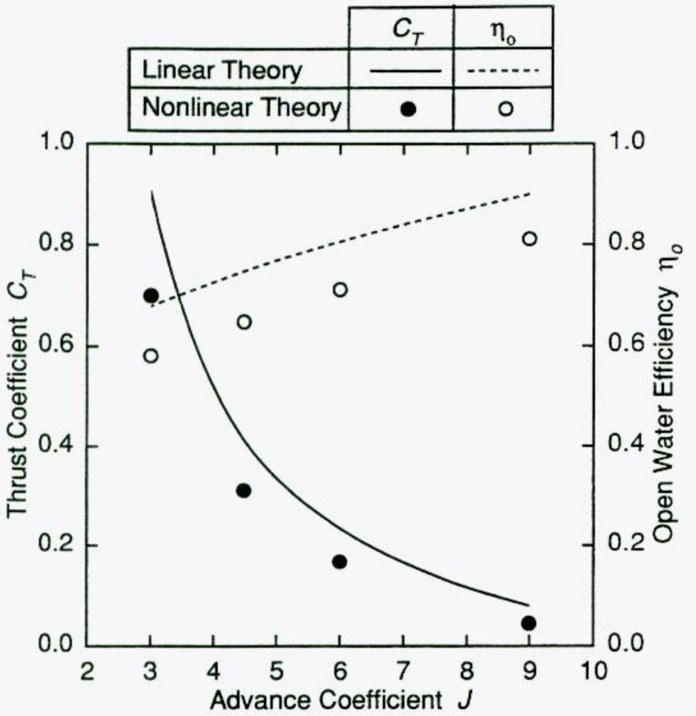

Fig. 5 Comparison of thrust coefficient and open water efficiency between linear (Kudo et al., 1984) and nonlinear (Kubota et al., 1984) theories; Two-dimensional, Section shape = NACA63A015, Rear half flexible, $E^* = 120$, $h/C = 0.6$, $\theta = 90$ deg., $\alpha = 10$ deg., Pitching axis = 25% chord station

resultant force vector into a direction more in-line with the direction of advance. As a result, a flexible foil becomes more efficient for the same level of thrust.

(2) The results calculated by the nonlinear theory agreed very well with the experimental ones if proper corrections were made for 3-dimensional and viscous effects. Figure 4 compares the thrust coefficient and open water efficiency between the corrected nonlinear theory results and an experiment made by Scherer (1968) for a rigid rectangular planform foil with an aspect ratio of 3.

(3) Although the linear theory overpredicted thrust, efficiency and power, it correctly predicted the overall trends as shown in Figure 5. The linear theory can therefore be used for various qualitative analyses.

(4) A nondimensional parameter for the foil motion, ε, Lighthill's feathering parameter (1970), was found suitable for analyzing the propulsor performance through an investigation on linear theory. The thrust and power (normalized by the heaving amplitude) and the efficiency were dominated by 3 foil motion parameters: the reduced frequency, the phase difference θ between heaving and pitching, and the feathering parameter ε. Design charts called θ–ε charts were drawn for various reduced frequencies. Using these charts, one can find the most efficient foil motion for a given thrust (or power), frequency and heaving amplitude.

Later, the Research Panel No.200-13 of the Shipbuilding Research Association of Japan (1986) investigated various ship propulsion devices. As a part of its investigation, hydrodynamic designs of a contrarotating propeller and an oscillating foil propulsor were made for a 200,000 DWT tanker. This ship was one that had been previously investigated by the Research Panel No.98 of the Shipbuilding Research Association of Japan (1967). The design point of the propulsors was selected as a ship speed of 14 knots. A 15% sea margin was applied. For comparison, a conventional screw propeller was also designed by a person in a shipbuilding company. He designed a 5 bladed MAU propeller of 8.4 m in diameter. The principal particulars of the ship and the propeller are shown in Table 2.

The principal foil particulars are shown in Table 3. Although it is known that a swept planform is more efficient (Liu and Bose, 1993), a rectangular planform is adopted here because the three-dimensionality correction method used by Kubota et al. (1984) was only valid for a

Table 2 Principal particulars of selected ship and propeller designed for comparison

Ship = 200,000DWT Tanker	
Length between Perpendiculars	300.00 m
Beam	54.5 m
Draft	17.825 m
Full Load Displacement	239,914 ton
Block Coefficient Cb	0.804
Propeller = MAU5-55	
Diameter	8.400 m
Pitch Ratio	0.722
Expanded Area Ratio	0.550
Blade Thickness Ratio	0.05
Number of Blades	5

Table 3 Principal particulars of the foil

Foil Section Shape	NACA 63A015
Foil Planform	Rectangular
Elastic Part	Midchord - T.E.
Foil Chord Length (C)	7.0 m
Foil Span Width (S)	49.0 m
Young's Modulus of Elastic Part	3.00×10^6Pa
Nondimensional Young's Modulus of Elastic Part	112.96 at Ship Speed = 14kt.

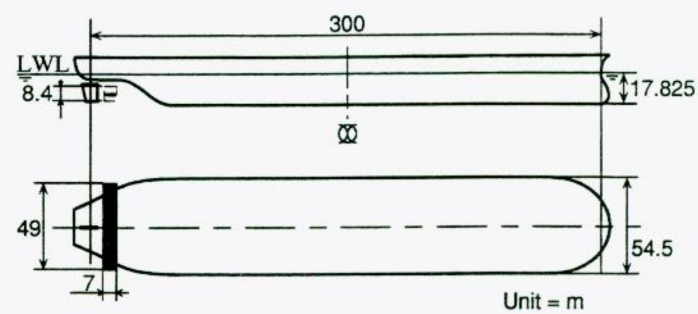

Fig. 6 Arrangement of ship and oscillating foil propulsor

rectangular planform. The parameters shown in Table 3 were determined as follows:

(1) Increase in the foil heaving amplitude raises the efficiency because of the increase in swept working area. On the other hand, the heaving amplitude is restricted by the ship draft in a similar manner to the case of a screw propeller. Therefore the double amplitude of heaving was set equal to the diameter of the screw propeller, i.e. 8.4 m. Thus, $h = 8.4/2 = 4.2$ m.

(2) The ratio of the heaving amplitude to the foil chord length, h/C, was set equal to that in the previous study (Kubota et al., 1984): i.e. 0.6. Thus, C = 7.0 m.

(3) The foil aspect ratio was set at 7 due to the restriction of the ship beam. Thus, S = 49.0 m, which is 0.9 times the ship beam.

(4) The foil section shape was NACA 63A015 (Abbott and von Doenhoff, 1959) which was the same as that examined in the previous work (Kubota et al., 1984). Similarly, the flexible part was assumed to be from the midchord to the trailing edge.

(5) The Young's modulus of the flexible part was set as $E = 3.00 \times 10^6$ Pa. This value can be realized by a rubber. The corresponding nondimensional Young's modulus at the ship speed of 14 knots becomes $E^* = 112.96$.

Figure 6 shows the arrangement of the foil and ship.

Linear theory was adopted to find an optimal foil motion: i.e. pitching amplitude and phase delay angle between pitching and heaving. Using the same EHP at a ship speed of 14 knots as assumed in the design of the screw propeller, the thrust coefficient was obtained from the following equation by assuming (1-t)=(1-w)=1: i.e. no interaction is assumed between the ship and

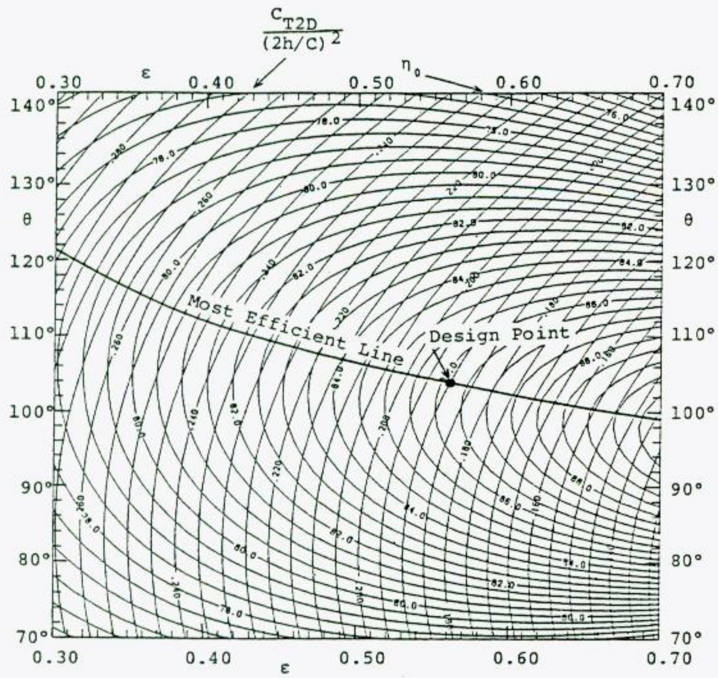

Fig. 7 Design chart obtained from linear theory; Rear half flexible foil, J = 6.173, E* = 112.96

Table 4 Optimum foil motion conditions obtained from linear theory

	Rigid Foil	Rear Half Elastic Foil
Heaving Amplitude (h)	4.2 m	
Piching Amplitude (α)	15.8 deg.	19.7 deg.
Phase Delay Angle of Pitching to Heaving (θ)	105.0 deg.	104.0 deg.

propulsor (Yamaguchi, 1992).

$$C_T = \frac{(EHP\times 75)/V_S(1-t)}{\left[(1/2)\times\rho\times\{(1-w)\times V_S\}^2\right]\times C\times S}$$

$$= 0.1482.$$

This is the value for "no sea margin". Strictly speaking, a sea margin should be considered on power. The "15% sea margin" here, however, is simply taken into account by multiplying this value by 1.15. Thus, the C_T including a 15% sea margin was estimated as

$$C_T = 0.1704.$$

The linear theory, which was used to produce the design chart, is for two-dimensional and inviscid flow. Therefore, the above value should be further corrected for three-dimensional and viscous effects. The three-dimensional correction method used by Kubota et al. (1984) requires knowledge of the foil advance coefficient J. The foil frequencies were assumed as N = 8.5 c.p.m. (J = 7.263) for the rigid foil and N = 10.0 c.p.m. (J = 6.173) for the rear half flexible foil. It should be noted that these frequencies were rough values used for the optimization using linear theory. They were then corrected through use of the nonlinear theory calculations to produce the required thrust. These values of advance coefficient were used to predict the three-dimensional and the viscous effects. A viscous drag coefficient of 0.01 was assumed. As a result the C_T values corresponding to the two-dimensional calculations were C_{T2D} = 0.1905 for the rigid foil and C_{T2D} = 0.1887 for the rear half flexible foil.

The comparison in Kubota et al. (1984) has shown that the linear theory predicts larger thrust than the more precise nonlinear theory (Figure 5). Therefore, the final required C_{T2D} values for the optimization were increased by 35% and 45% respectively to account for this. Hence,

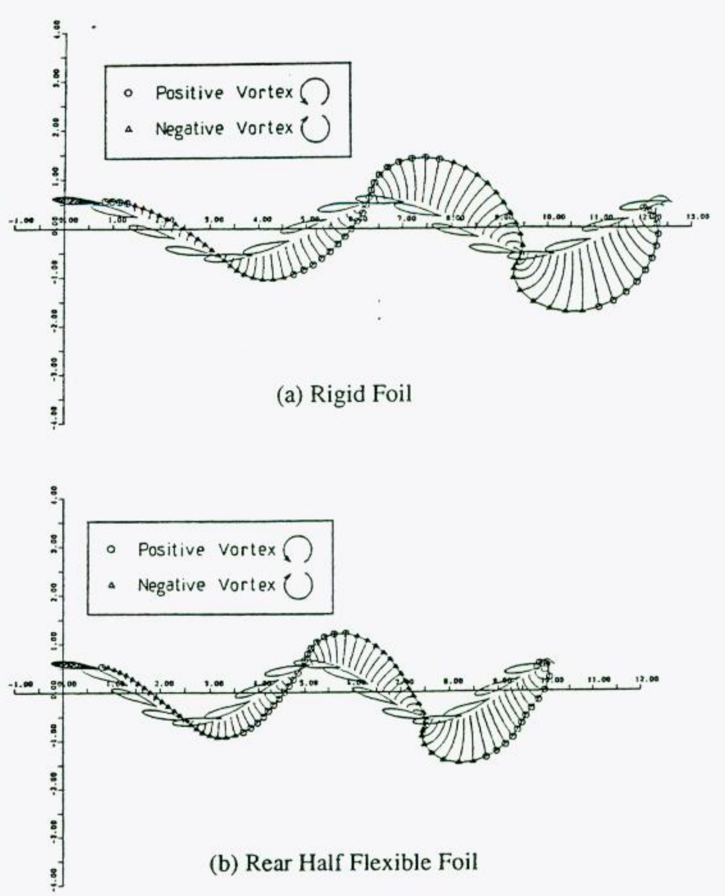

Fig. 8 Appearance of foil motion and trailing vortices at the design point (ship speed = 14 knots, 15% sea margin); N = 9.85 c.p.m. (J = 6.27) for the rigid foil, N = 12.34 c.p.m. (J = 5.00) for the rear half flexible foil

For the Rigid Foil: $C_{T2D} = 0.2572$ ($C_{T2D}/(2h/C)^2 = 0.1786$)
For the Rear Half Flexible Foil: $C_{T2D} = 0.2736$ ($C_{T2D}/(2h/C)^2 = 0.1900$).

Figure 7 shows the design chart obtained from the linear theory for a rear half flexible foil. This is a contour map of thrust coefficient $C_{T2D}/(2h/C)^2$ and efficiency η_o. Also drawn is the most efficient line for a given thrust (shown by a thick line). The most efficient (ε,θ) combination can be read as the crossing point of the most efficient line and the curve of the above obtained $C_{T2D}/(2h/C)^2$ value (= 0.1900). This is denoted by a solid circle in the figure. The same procedure was carried out for the rigid foil. Table 4 shows the optimum foil motion conditions for the respective foils. The optimum foil motion condition changes in accordance with the ship speed. However, for simplicity in this paper the propulsor hydrodynamic performance is calculated with the foil motion conditions unchanged.

Finally the nonlinear theory was used to predict the propulsive performance of the designed oscillating propulsors with viscous and three-dimensional corrections taken into account. Figure 8 illustrates the foil motion and the trailing vortices at the design point. Although one might imagine a high frequency motion from the words "oscillating foil", for a high efficiency the oscillation is in fact a relatively low frequency. Figure 9 compares the three propulsors, rigid and flexible foils and screw propeller, and shows quasi-propulsive efficiency (van Manen and van Oossanen, 1988) against speed for this ship. The screw propeller operates over a relatively small swept area which lies well within the ship wake and is able to produce the required thrust with a lower power input than it would in unrestricted flow. As mentioned before, negligible interaction was assumed between the hull and the oscillating propulsors: open and quasi-propulsive efficiencies are assumed to be the same for these propulsors. Thus although the open water efficiencies at the design condition of the two oscillating propulsors are larger than those for the screw propeller by 17-25%, when placed behind the ship, the quasi-propulsive efficiency of the screw propeller becomes greater than that of the rigid foil by about 3-4%. The partly flexible foil is

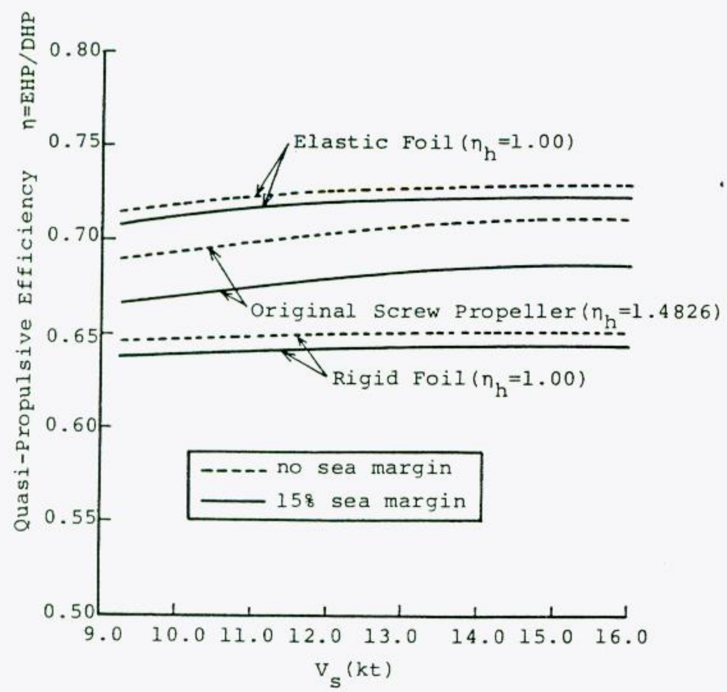

Fig. 9 Comparison of quasi-propulsive efficiency between flexible and rigid foils and a MAU5-55 conventional screw propeller; The hull efficiency η_h is assumed as 1.0 for the oscillating foils, viz. no interaction between the hull and propulsor is assumed for the oscillating foils.

(1)	Ideal Efficiency Obtained from Momentum Theory	
(2)	Open Water Efficiency	Screw Propeller (MAU5-55)
(1) - (3) (3) - (4) (4)	Loss due to Fluid Lateral Motion Loss due to Viscosity Open Water Efficiency	Rigid Foil
(1) - (5) (5) - (6) (6)	Loss due to Fluid Lateral Motion Loss due to Viscosity Open Water Efficiency	Rear Half Flexible Foil

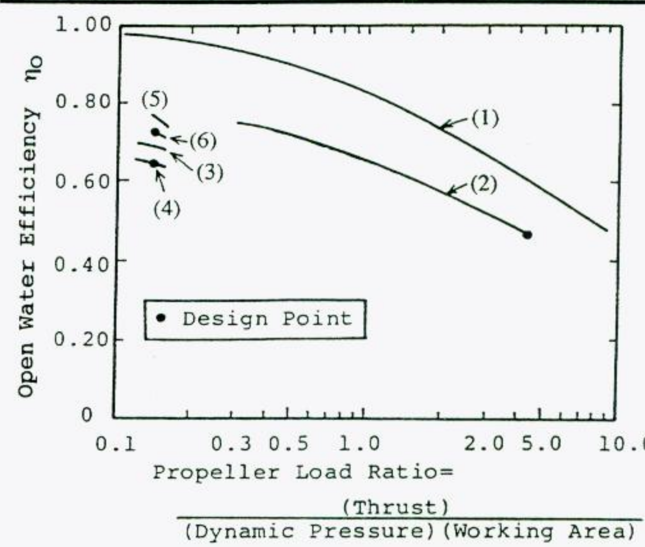

Fig. 10 Comparison of open water efficiency as a function of propeller load ratio

about 8% more efficient than the rigid foil. The propulsive efficiency of the partly flexible foil is about 72% and is higher than that of the screw propeller by about a 3.5% difference or 5.0% in ratio. The ship speed affects the flexible foil efficiency more than the rigid foil one. This is because lower ship speed increases the nondimensional Young's modulus E^*, which means the flexible part becomes stiffer for a reduction in the dynamic pressure. It is also seen that the efficiency of the oscillating propulsors is less sensitive to the variation of ship speed and sea margin than that for the screw propeller. This is one of the advantages of an oscillating propulsor: the changes in the working conditions affect the efficiency less than for a screw propeller.

Different propulsors may be compared on a basis of open water efficiency at varied propeller load ratio (van Manen, 1973) (propeller load ratio = $C_T/(2h/C)$ = (Thrust)/$\{(\rho V_a^2/2)\times A\}$, where A is the swept area of the propulsor in a plane perpendicular to the direction of advance). Figure 10 shows the comparison between the maximum theoretical or ideal efficiency from propeller momentum theory (van Manen and van Oossanen, 1988), the conventional screw propeller and the oscillating propulsors. At the design point, the propeller load ratio of the oscillating propulsors is much lower than that for the screw propeller and the efficiency increase for the oscillating propulsors is primarily due to the reduction in load ratio caused by the increase in swept area of the device (by a factor of 7.4 here; a design possible because of the wide ship beam). The losses due to wasted lateral water motion (here, "lateral" means vertical and horizontal directions.) are much larger for the oscillating propulsors and this means that the oscillating propulsors are less efficient than a screw propeller if they are compared under conditions of similar load ratio. Chordwise flexibility reduces these losses and further improvement in efficiency is expected from use of the swept back planforms (van Dam, 1987; Liu and Bose, 1993) and perhaps the spanwise flexibility found in naturally evolved oscillating propulsors. In addition, horizontally oriented foils are known as effective wave energy absorbers (Wu, 1972; Isshiki and Murakami, 1982-1984; Isshiki and Murakami, 1986; Bose and Lien, 1990; Lai et al., 1993) which in favorable conditions would increase propulsive effectiveness. The practical application of oscillating propulsors for ship propulsion has yet to be demonstrated, although a propulsor of this type has been used to propel a small boat (Isshiki et al., 1987).

CONCLUSIONS

An oscillating propulsor with a rear half flexible foil revealed 5% higher propulsive efficiency than a comparable screw propeller primarily because of an increase in swept working area. Thus oscillating foil propulsion becomes effective for a wide beam ship where high propeller loading is unavoidable for a conventional single screw propeller. In the present work, only a rectangular planform was investigated because of the limitation of the theories used. Further increase in efficiency might be realized by crescent planforms with spanwise flexibility like an aquatic animal's fin.

Needless to say, still left are significant problems concerned with the development of suitable driving devices and mechanisms. From a hydrodynamical aspects, it is desirable to investigate the interaction between an oscillating propulsor and the hull, and to develop a hull form suitable for oscillating propulsion. A narrow wake is disadvantageous to an oscillating propulsor because an oscillating propulsor would span the wake and the undisturbed water away from the hull. Also, the losses due to lateral fluid motion are significant. If these are taken into consideration, promising results might be obtained with a hull form like a transom stern as shown in Figure 6. The hull bottom above the foil might suppress the vertical fluid motions, resulting in considerably higher efficiency. Such a mirror effect has been shown to reduce heave reactions in a small boat with an oscillating propulsor (Isshiki et al., 1987).

ACKNOWLEDGMENTS

The authors thank Prof. H. Kato, University of Tokyo, Dr. H. Isshiki, Hitachi Zosen Inc. and other members of the Shipbuilding Research Association Research Panel No. 200-13 for their advice. The authors gratitude is extended to the Kajima Foundation, the Institute for Marine Dynamics, National Research Council of Canada and the Natural Sciences and Engineering Research Council of Canada for their support.

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